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Vibration dynamics of a commercial vehicle engine suspended on adaptronic mounting system

Hoda Yarmohamadi and Viktor Berbyuk

Abstract In this paper two computational models for commercial vehicle engine vibration dynamics are presented. The first model is conventional elastomeric engine mounting system comprising elastic, viscous and friction functional components which expresses the nonlinear behaviour of the dynamic stiffness and damping of the engine mounts as functions of both the frequency and amplitude of excitation. The second model is based on adding actuators, sensors and a controller to the elastomeric mounts making the engine mounting system an adaptronic one. Parameter optimization of the proposed adaptronic engine mounting system has been done. Computational models are implemented in Matlab /Simulink with GUI suitable for engine vibration dynamics analysis under different engine excitations and realistic road inputs. Simulations of engine vibration and transmitted forces to the vehicle structure are presented which are obtained by using both computational models for the same inputs of engine and road excitations. Comparison of the obtained results for adaptronic mounting system (AMS) to the corresponding results of conventional mounting system (CMS) demonstrates up to %24 reduction in the magnitude of the transmitted force to the vehicle structure achieved with adaptronic engine mounting system that includes two actuators with forces not exceeding 500(N).

1 Introduction

The vehicle engine mounting system is a vibration control system that can be passive, semi-active or active. It isolates the noise and vibration and also the transmitted forces to the vehicle structure. At present conventional rubber-metal mounts are widely used for commercial vehicle engine mounting systems. However, these elastomeric mounts can not ideally fulfil the contradictory characteristics of de-

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sired stiffness and damping for both engine and road excitations. Therefore, a lot of research is going on for semi-active and active engine mounting systems [7]. In paper [5] a hydraulic active engine mount for commercial vehicles was studied.

In the present work, vibration dynamics of engine suspended on conventional and adaptronic mounting systems is analyzed and reduction in the transmitted forces to chassis is obtained by adding two actuators with forces less than 500(N).

2 Modelling

In this study conventional and adaptronic mounting systems are considered. Adaptronic mounting system is made by adding actuators to conventional mounts. The commercial vehicle engine is modelled as a rigid body with mass M_e connected with engine mounting system to the flexible and moving chassis as shown in Fig. 1. $O_1x_1y_1z_1$ represents the global reference frame and $Oxyz$ is the body fixed reference frame with its origin at engine centre of gravity. The x-axis is taken parallel to the crank shaft and z-axis is in the vertical direction when in static position.

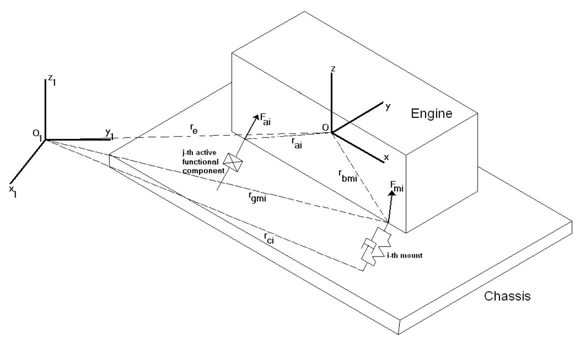

Fig. 1 Engine suspended on mounting system [1]

The positions of the connection points of ith conventional mount and jth active functional component to the engine are denoted by $\mathbf{r}_{mi} = \left[x_{mi}, y_{mi}, z_{mi}\right]^T$ and $\mathbf{r}_{aj} = \left[x_{aj}, y_{aj}, z_{aj}\right]^T$ with respect to body-fixed reference frame, respectively. Moreover, $\mathbf{r}_e = \left[x_e, y_e, z_e\right]^T$ is the position vector of the engine centre of mass with respect to global coordinate system.

Rotational displacements of the mounts are neglected and only translational displacements are taken into account. The engine equations of motion are written using the Newton-Euler approach. The rotation of the body is assumed to be small, therefore the translational and rotational equations of motion of the engine can be written as in [3]

$$M_e \ddot{x}_e = \sum F_x, \quad M_e \ddot{y}_e = \sum F_y, \quad M_e \ddot{z}_e = \sum F_z \tag{1}$$

$$\begin{aligned} I_{ex}\ddot{\theta}_{ex} &= \left(I_{ey} - I_{ez}\right)\dot{\theta}_{ey}\dot{\theta}_{ez} + \sum T_x, \quad I_{ey}\ddot{\theta}_{ey} = \left(I_{ez} - I_{ex}\right)\dot{\theta}_{ez}\dot{\theta}_{ex} + \sum T_y, \\ I_{ez}\ddot{\theta}_{ez} &= \left(I_{ex} - I_{ey}\right)\dot{\theta}_{ex}\dot{\theta}_{ey} + \sum T_z \end{aligned} \tag{2}$$

where $\theta_{ex}, \theta_{ey}, \theta_{ez}$ are the rotations of the engine about x, y, and z axis of the global reference frame, respectively. $\sum F_x, \sum F_y, \sum F_z$ are the sums of all forces acting on the engine in the x, y, and z direction, respectively. Moreover, $\sum T_x, \sum T_y, \sum T_z$ are the sums of all torques that act on the engine.

When the mounting system is adaptronic, actuator forces, F_{aj}, appear in the sum of forces and torques in the equations of motion.

$$\mathbf{F} = \mathbf{F}_{conventional} + \mathbf{F}_e + \sum_{j=1}^{m} \begin{bmatrix} F_{aj} \cos\alpha_j \cos\beta_j \\ F_{aj} \cos\alpha_j \sin\beta_j \\ F_{aj} \sin\alpha_j \end{bmatrix} \tag{3}$$

where $\mathbf{F}_{conventional}$ is the resulting force from the conventional mounts and $\mathbf{F}_e$ is the resulting force from the engine. α and β demonstrate the inclination of the actuator in space. Total torques acting on the engine with respect to global reference frame are

$$\begin{aligned} T_x &= \sum_{i=1}^{n}\left(F_{mzi} y_{mi} - F_{myi} z_{mi}\right) + T_{ex} + \sum_{j=1}^{m} T_{axj} \\ T_y &= \sum_{i=1}^{n}\left(F_{mxi} z_{mi} - F_{mzi} x_{mi}\right) + T_{ey} + \sum_{j=1}^{m} T_{ayj} \\ T_z &= \sum_{i=1}^{n}\left(F_{myi} x_{mi} - F_{mxi} y_{mi}\right) + T_{ez} + \sum_{j=1}^{m} T_{azj} \end{aligned} \tag{4}$$

where F_{mx}, F_{my}, F_{mz} are the reaction forces of the mounts w.r.t. body-fixed reference frame. T_{ex}, T_{ey}, T_{ez} are the components of the resultant torque from engine excitations and $T_{axj}, T_{ayj}, T_{azj}$ are the actuator torque components around the axes.

The transmitted force to chassis in each direction (x, y, z) is calculated as the sum of the forces transmitted to the frame through engine mounts and active functional components in that direction. The total transmitted force, F_T, is the magnitude of the resulting force at each instant of time as stated in equation (5).

$$F_T = \sqrt{F_{Tx}^{\ 2} + F_{Ty}^{\ 2} + F_{Tz}^{\ 2}} \tag{5}$$

2.1 Conventional mounting system

Usually engine mounts are represented with a linear parallel spring and damper in vehicle models but this model does not take care of the nonlinear behaviour of the mount, e.g. due to friction phenomenon. Also, with constant spring and damper coefficients, the model does not show amplitude dependency of the stiffness and damping of the mount. Our methodology of modelling is to put functional components of the elastic, viscous and friction behaviours together. By identifying model parameters through optimization with LMS algorithm using measurement data for real rubber-metal commercial vehicle engine mounts, nonlinear stiffness and damping of the mounts are calculated as functions of frequency and amplitude of excitation.

Elastic part is a linear spring and viscous part consists of two sets of series linear spring and damper put together in parallel. Friction is defined using the model proposed in [2]. The sketch of the conventional mount model is shown in Fig. 2. The model is one-dimensional and the relation between force and displacement is based on the superposition of elastic, viscous and friction forces.

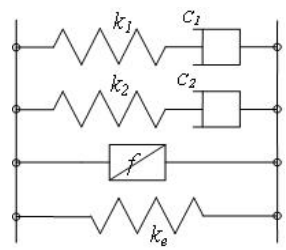

Fig. 2 Sketch of the conventional mount model using elastic, viscous and friction functional components

For harmonic excitation input to the mount, $x = x_0 \sin(\omega t)$, where x_0, ω and t are displacement amplitude, excitation frequency and time respectively, the stiffness and damping of the conventional mounts are computed as follows

$$S = F_0 / x_0 , \quad D = E / F_0 x_0 \tag{6}$$

where F_0 is the total steady state force amplitude and E is the total energy loss per cycle. D is a non-dimensional damping indicating how much energy from the total energy input has been dissipated.

In Fig. 3 some results obtained with the developed computational model are presented along with the corresponding measurement data for a commercial vehicle engine mount. As can be seen in the plots, the model predicts the stiffness and damping of the mount as a function of amplitude and frequency of excitation with a good agreement to measurement data [6].

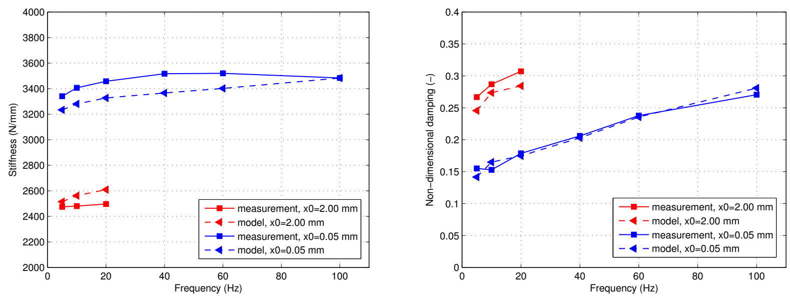

Fig. 3 Stiffness and damping of the mount for two different amplitudes

To analyze the engine vibration dynamics on conventional mounts, equations (1)-(5) are used and all the forces and torques concerning the active functional components are put to zero.

2.2 Adaptronic mounting system

Actuators are added to the conventional mounting system, making it an adaptronic one. According to conventional mount deflections and transmitted forces to chassis under different road excitations, it can be seen that deflections of the front mount in lateral direction and the rear mounts in vertical direction are the largest and most critical ones (see Table 2 and Table 3). Therefore, two actuators are added to the conventional mounting system, one in front in lateral direction and the other one in rear between the two rubber-metal mounts in vertical direction.

The actuator force is modelled as a function of displacement and velocity with linear and nonlinear elastic and viscous components as stated below

$$F_a = ax + bx^2 + c\dot{x} + d\dot{x}^2 \qquad (7)$$

To find the parameters of the actuator forces in equation (7), optimization is performed in which the cost function is the summation of the squared transmitted forces to the chassis over the simulation time as also used in [4]. The optimization is constrained with limitations on the lower and upper bounds of the parameters so that the force exerted by the actuator does not exceed 500(N). For our purpose, Matlab subroutine *fmincon* is used. Equations (1)-(5) are used for transmitted force and engine vibrations calculation.

The control system is feedback control and practically, we need accelerometers at connection points of mounts to engine and chassis to get information about the mounts and input to the controller. The controller must respond fast and excite the actuators.

3 Simulation, analysis and comparison

In this section results of simulations are presented and analyzed. The vibration dynamics of the commercial vehicle engine mounted on conventional and adaptronic mounting systems is investigated under different dynamic and kinematic excitations. The source of the dynamic excitation is engine internal processes and that of kinematic excitation is the road surface irregularities.

Inputs to the model

To investigate the engine vibration dynamics, simulations are performed using the data of 6 cylinder commercial vehicle engine with mass of 1900(kg). Inertia properties of the engine, locations and characteristics of the conventional mounts which are used in simulations can be obtained from Volvo 3P, Gothenburg, Sweden.

For dynamic excitation, small amplitude-high frequency excitation is considered. Kinematic excitation, the displacement of the chassis at each mount connection point to the chassis, is applied to the mounts. Two realistic road models from Volvo 3P are used in the simulations, good and bad roads with constant vehicle velocity of 21.9(m/s), explained in Table 1.

Table 1 Kinematic excitation characteristics for good and bad roads

Mount	Min displacement x, y, z (mm)	Max displacement x, y, z (mm)
Front mount-good road	-0.6076, -0.7170, -4.4893	0.6652, 0.0294, 3.7586
Rear left mount-good road	-0.4168, -0.5229, -3.7102	0.4046, 0.0518, 2.5322
Rear right mount-good road	-0.4789, -0.5151,-3.6927	0.4024, 0.0512, 2.5685
Front mount-bad road	-2.7757, -2.9414, -21.6731	2.5020, 0.0373, 18.2192
Rear left mount-bad road	-1.8397, -2.0417, -18.3614	1.7132, 0.1629, 14.9999
Rear right mount-bad road	-1.8705, -2.0653, -18.1554	1.6772, 0.2556, 14.8079

Outputs of the model

In order to compare the conventional and adaptronic mounting systems together, translational and rotational displacements and accelerations of the engine, mount deflections and forces, actuator forces, as well as the transmitted forces to chassis are obtained from simulations over 10 seconds.

3.1 Transmitted forces to the chassis

The simulation results for the transmitted forces to the chassis for good and bad road are shown in the following figures.

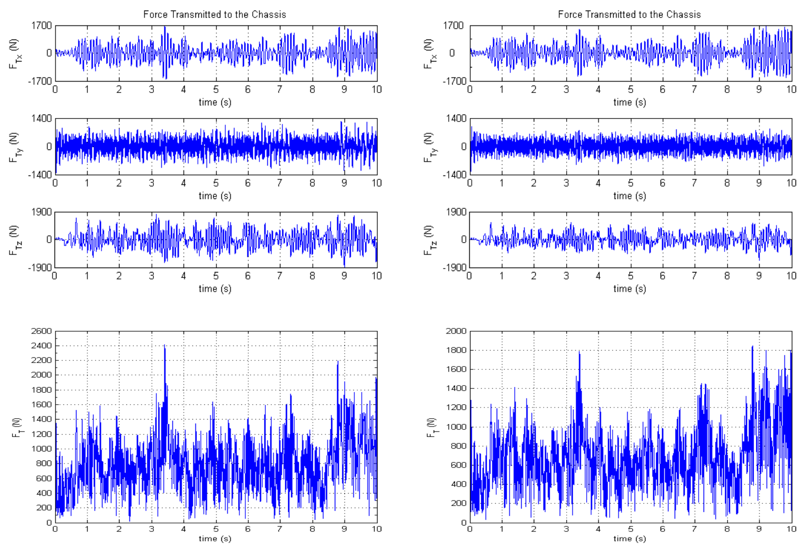

Fig. 4 Transmitted force to chassis on good road: (left) CMS, (right) AMS

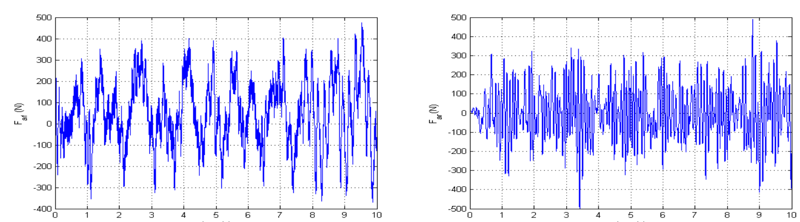

Fig. 5 Actuator forces on good road: (left) lateral front actuator, (right) vertical rear actuator

As it follows from Fig. 4 with the proposed adaptronic mounting system, the total transmitted forces to the chassis have been reduced. The maximum value has decreased from 2416(N) by %24 to 1848(N). The RMS value of the total transmitted force improved %14 by decreasing from 891(N) to 768(N) with adaptronic engine suspension for which the actuator forces do not exceed 500(N).

For the bad road conditions, with the proposed adaptronic mounting system and actuator forces less than 500(N), the total transmitted forces are reduced to some extent (See Fig. 6). The maximum total transmitted force is reduced from 6778(N) to 6660(N) which is almost %2. The RMS value of the total transmitted force is decreased from 2748(N) for CMS to 2589(N) for AMS which implies %6 improvement.

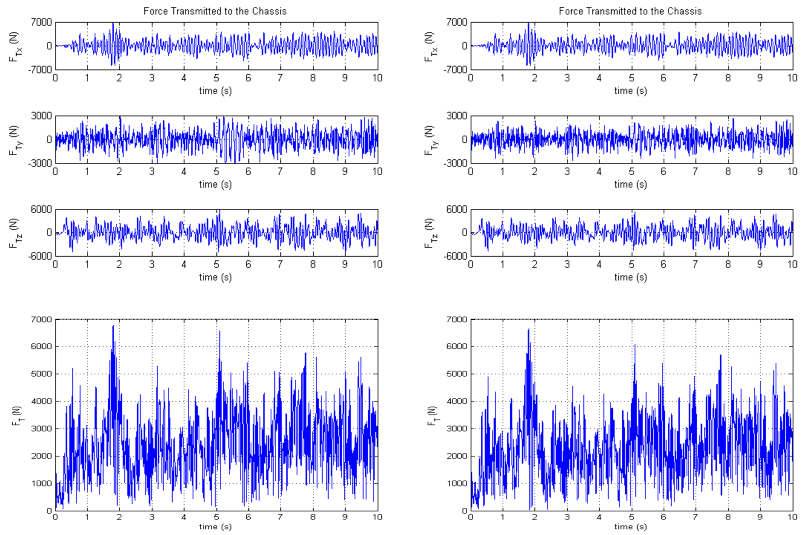

Fig. 6 Transmitted force to chassis on bad road: (left) CMS, (right) AMS

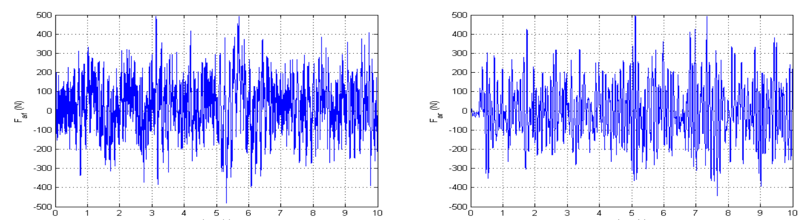

Fig. 7 Actuator forces on bad road: (left) lateral front actuator, (right) vertical rear actuator

3.2 Engine and mount displacements

Engine displacement and acceleration are not affected much with the adaptronic engine mounting system compared to the conventional one. It might be so since the actuator forces used are small compared to the weight of the engine. In Fig. 8 the translational and rotational displacements of the engine on bad road are presented for adaptronic mounting system as an example. In Table 2 and Table 3, minimum and maximum deflections on the mounts during simulation time for good and bad roads are presented, respectively. Similar to what we obtained for the engine displacements, the mount deflections are not that much affected by using adaptronic mounting system instead of the conventional one.

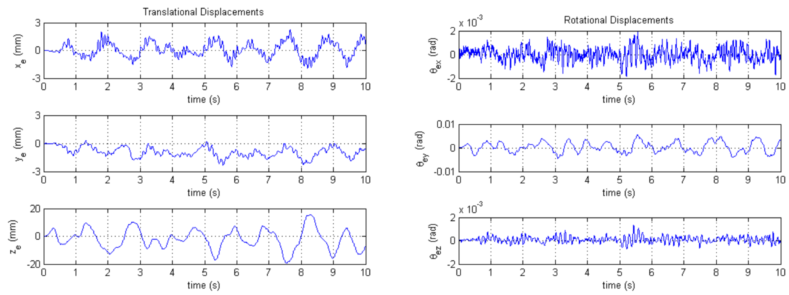

Fig. 8 Translational and rotational engine displacements suspended on AMS on bad road

Table 2 Minimum and maximum deflections on the mounts for good road

Mount	Min deflection x, y, z (mm)	Max deflection x, y, z (mm)
Front mount-conventional	-0.1657, -0.2727, -0.1735	0.1619, 0.3607, 0.1784
Rear left mount-conventional	-0.1940,-0.0959, -0.2800	0.2182, 0.0837, 0.2302
Rear right mount-conventional	-0.2068, -0.0888, -0.3332	0.2158, 0.0949, 0.3492
Front mount-adaptronic	-0.1664, -0.2512, -0.1833	0.1545, 0.3985, 0.1786
Rear left mount-adaptronic	-0.1938,-0.0878, -0.3448	0.2241, 0.0889, 0.2701
Rear right mount-adaptronic	-0.2243, -0.0898, -0.2867	0.1901, 0.0857, 0.2958

Table 3 Minimum and maximum deflections on the mounts for bad road

Mount	Min deflection x, y, z (mm)	Max deflection x, y, z (mm)
Front mount-conventional	-0.6753, -1.5165, -0.6638	0.6345, 1.4627, 0.6633
Rear left mount-conventional	-0.7072,-0.2647, -0.8489	0.8890, 0.2536, 0.6808
Rear right mount-conventional	-0.9124, -0.2792, -0.7983	0.8334, 0.2893, 1.0241
Front mount-adaptronic	-0.6551, -0.9895, -0.6524	0.6052, 1.2128, 0.6276
Rear left mount-adaptronic	-0.7756, -0.2534, -0.6508	0.8793, 0.2736, 0.6699
Rear right mount-adaptronic	-0.8489, -0.2328, -0.6251	0.8009, 0.2435, 0.7402

4 Conclusions

In the automotive industry, there is interest in more advanced (semi-active/active) engine mounts due to their capabilities of fulfilling contradictory requirements of the stiffness and damping of the mounting system for road and engine excitations. To improve existing and develop new more efficient engine mounting systems, an accurate model of the conventional elastomeric mounts is desirable. In this study, a computational model consisting elastic, viscous and friction functional components is presented. Analysis of the model has shown that this model express the nonlinear behaviour of the dynamic stiffness and damping of the engine mounts as a function of frequency and amplitude of excitation. Furthermore, an adaptronic mounting system is developed by adding two actuators, with maximum capacity of 500(N), to the conventional mounting system and its mathematical model is presented.

Computational models are implemented in Matlab/Simulink. Simulations of engine vibration and transmitted forces to the vehicle structure are conducted for engine and realistic road excitation inputs.

Comparison of the obtained results for conventional and adaptronic mounting systems for the same inputs of road and engine excitations demonstrates valuable improvement in commercial vehicle engine vibration isolation and reduction of the transmitted forces to the vehicle structure achieved with adaptronic engine mounting system. During the simulation time, the maximum value of the total transmitted forces to the chassis has decreased by %24 and %2 while the RMS value of the transmitted forces has decreased by %14 and %6 for good and bad road inputs, respectively. However, due to small actuator forces compared to the engine weight, 1900(kg), the deflections of the rubber-metal mounts and the displacement of the engine is not affected much by changing the engine mounting system from conventional to adaptronic. By increasing the actuator force limit, more improvements can be achieved for the transmitted force reduction to chassis.

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